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TEST STAND OF OSCILLATIONS OF TURBIN ROTORS IN BEARINGS WITH GAS HYDRODYNAMIC LUBRICATION

by

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13. ABSTRACT

A test stand was designed for investigations of rotor vibrations caused by a misalignment, by resonant phenomena, and by the loss of stability in the lubricating layer. Anti-friction bearings and different slide bearings with an oil or gas lubricant can be used as rotor supports. One support of the test stand is adjustable, and a set of rotors with detachable disks is provided. The stand is suitable for studying rotors up to 1240 mm long and 350 mm in diameter. It can also be used to study vibrations caused by misalignment of the bearings, and to study the external damping, the centering of bushings, and the possibility of modeling the turbing flow passage. [AR9010718]

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Turbine Rotor						
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Antifriction Bearing						
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Organic Lubricant						

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By: A. V. Palladiy

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TEST STAND FOR STUDY OF OSCILLATIONS OF TURBIN ROTORS
IN BEARINGS WITH GAS HYDRODYNAMIC LUBRICATION

[Article by A. V. Palladiy, G. A. Pospelov, Ye. I. Spektor; Russian, Gazovaya Smazka, Podshipnikov, No 9, 1968, p 283-289]

As we know, the motion of rotors of high speed turbins, turning in friction bearings, can be accompanied by strong vibrations. The basic causes for the vibrations include the influence of residual imbalance, resonance factors, losses of stability of motion resulting from the influence of the lubricant layer, and a number of other factors.

In spite of the large number of theoretical and experimental works, no reliable engineering methods have yet been developed for planning calculation of the vibration of rotors in hydrodynamic and gas dynamic bearings. Theoretical studies have not yet succeeded in considering all factots causing vibrations, and it is necessary to analyze a simplified rotor-bearing model. For example, works [1, 2] do not consider the influence of the lubricant layer completely, while works [3, 4] study only models of a balanced rotor. Therefore, the results produced are only qualitative in nature. A comparison of the available quantative recommendations for determination, for example, of the zones of vibration of flexible rotors indicate their mutual incompatibility and failure to correspond to operational data [5].

The unreliability of the method suggested for calculation of rotor vibrations also result from the fact that the available experimental data cover only certain particular cases and cannot serve as a basis for the development of a general method of calculation. Experiments with flexible rotors in gas dynamic bearings have not been performed at all. Systematic data are not available on the influence of gas dynamic flow in turbines on the stability and vibration of the rotor-bearing system.

For this reason, a universal test stand was created for the investigation of the entire range of vibrations of turbine rotors.

The test stand design developed allows studies of the vibration of rotors caused by imbalance, resonance phenomena and losses of stability in the lubricant layer to be studied. The bearings supporting the rotor on the stand may be either rolling-surface bearings or various types of friction bearings with oil or gas lubrication. In order to study the influence of span length and distribution of masses along the length of the rotor on vibrations, the test stand has one moving bearing and a set of model rotors with seated discs. The test stand also allows the study of actual rotors up to 1240 mm in length with wheel diameters of up to 350 mm. The possibility is provided of studying the influence of noncoaxiality and skew of bearings, external damping and clutch centering on vibration, and allows the gas-flow portion of a turbine to be modeled. The test stand is rotated by a model P-91 direct current motor with a power of 55 kw. During tests of actual rotors, a P-101, 100 kw motor can be used. The electric motor is supplied with 220v dc by a 2-machine unit installed in a separate room. The rotating speed of the motor is adjusted using 2 rheostats, one of which is connected to the exciter winding of the dc generator, while the other is connected to the exciter winding of the motor itself. This allows smooth changing of the rotating speed of the experimental rotor between 3,000 and 25,000 rpm.

The universal stand is mounted on cast iron plates and consists (Figure 1) of electric motor 1, transmission 2, reducing gear 3 and test chamber 4. The motor, transmission and reducer are connected to each other by type MUVP dog clutches, the test chamber and reducer are connected by a toothed clutch.

Fig. 1. Installation Assembled.

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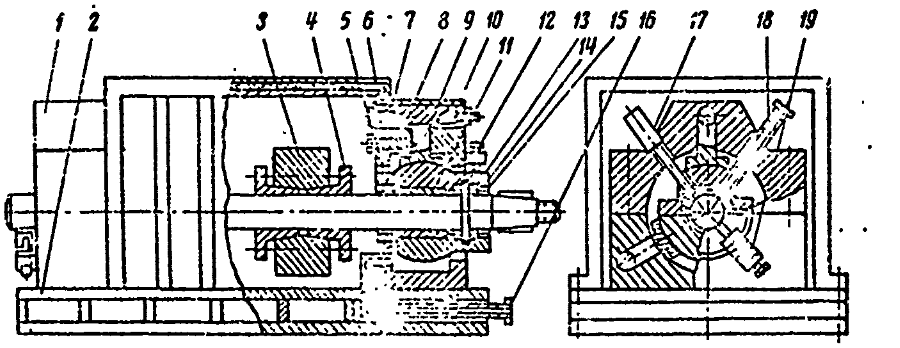

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Fig. 2. Test Chamber.

The test chamber (Fig. 2) consists of a welded base 2, attached to moving upright 1 and nonmoving upright 7. The rotor being tested is mounted in bearings 11 and covered with welded cover 6 with hollow walls filled with sand. The model rotors are made with journal diameters of 50, 60 and 80 mm. Each rotor is equipped with a set of discs 3. Using self-braking chuck 4, the discs can be attached to the shaft at any distance from the bearing.

Two flat, horizontal guides are attached to the base plate, along which moving upright 1 can be moved. There are 2 additional vertical guides in the base plate to prevent skew of the upright as it moves. The upright is connected to the guides by 4 bolts with primary heads, which move together with the uprights in a slot in the horizontal guides and are prevented from falling out or rotating by a special protective cap. Since the slots are milled into the plane of the horizontal guides, bending deformation of the upright relative to the base does not occur when the bolts are tightened. The maximum movement of the moving upright is 730 mm, corresponding to the change in distance between bearings with its limits of 200 to 930 mm.

The body of the upright consists of two halves and has a horizontal plane of separation. It contains 3 sliding blocks 9 at angles of 120°, used for installation and centering of the bearings. The contacting surfaces of sliding blocks 9 and bearing body 14 are spherically shaped, allowing the bearing to be rotated in any direction. The axes of the bearings are moved by radial movement of the sliding blocks by means of wedges 10. Since when the bearing is moved, the centers of the spheres of the body and the sliding blocks do not correspond, tightening of the wedges causes the blocks to skew in their directions within the limits of the clearance at that point. Bolts 8 and 12 are used to fix the sliding blocks and bearing body in place.

The uprights can hold various types of bearings: hydrodynamic, gas dynamic, gas static, etc. The non-moving upright contains a radial-thrust bearing, while the moving upright contains only a radial bearing. Figure 2 shows a radial-thrust hydrodynamic bearing. Bushing 11 of the radial bearing and thrust bearing 13 are placed in bearing body 14, which has a horizontal plane of dissembly. Oil is fed into the lubricating clearance through tube 18, and drains through tube 16. The bushing temperature is measured by thermometer 17. The bearing body also contains oil seals 5 and 15.

Figure 3 shows one version of a bearing with gas lubrication. Bearing body 2 is made of a single piece. Discs 5 and 7 of the thrust bearing and steel cover 6 are held to it by pins and bolts. The clearance in the thrust bearing is regulated by changing the thickness of the cover. The thrust bearing is a gas static bearing. Air is fed into the clearance through drilled apertures 8 and 9. The bushing of radial bearing 3 is installed in body 2 and compressed by nut 1. During starting and acceleration of the rotor, air is fed into the clearance through 2 rows of radial apertures 0.5 mm in diameter in order to decrease wear of the friction couple. The friction couple is selected considering the possibility of brief contact of journal and bushing during the process of rotation. The shaft is made of type 45 steel, hardened to HRC 40-45. The bushing is made of cast iron or type E carbon graphite.

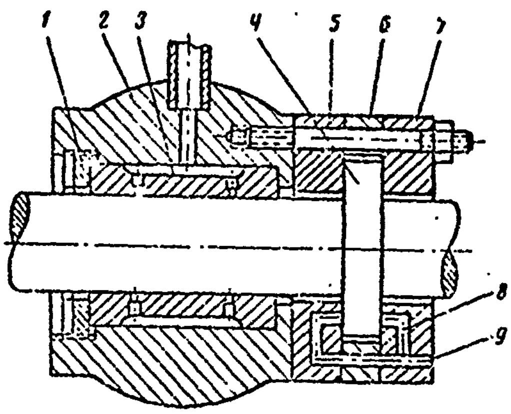

Fig. 3. Gas-lubricated Bearing.

There are two systems of air lines for air supply to the bearings: a low pressure system, up to 10 atm, and a high pressure system, up to 150 atm. In both cases the air is fed by type IK-65 or DK-2 compressors into air tanks, then from the tanks into the high or low pressure systems.

Electronic measuring apparatus is used to determine the qualitative and quantitative picture of rotor vibrations. Capacitive type sensors 19 are installed on each bearing (see Fig. 2) in planes perpendicular to the axis of the shaft and at 90° to each other. The plates of the sensors are ground to fit the shaft. As the position of the journal in the bearing changes during vibration, the sensor feeds a signal through a special amplifier to a cathode or loop oscilloscope. The 4 sensors, 2 on each bearing, indicate the vibrations of the shaft in both bearings at any moment in time, comparing them as to phase and amplitude.

Thus, this test stand allows various studies of rotor vibration in gas hydrodynamic bearings in various operating modes to be performed, with variable parameters of the rotor (rigidity, weight, number of discs, length of span between bearings, and bearings) diameter, clearance, L/d ratio).

We expect that the results of experiments using the test stand will allow new recommendations to be produced for the calculation of vibrations of rotors in gas hydrodynamic bearings.

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